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EXPERIMENTS WITH A FULLY INSTRUMENTED SPLIT STIRLING CRYOCOOLER

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A development program is being undertaken by L'Air Liquide in order to establish a practical model that can be used to accurately size and optimise split stirling cryocoolers.

For any given application the performance of this type of refrigerator must be carefully optimised and, in addition, the specifications for one application may very greatly from those needed for another (eg. operating temperature, cold power, cold finger volume, size and dead volume in the interconnecting line etc...). The final optimised design for any particular application requires time consuming and expensive testing as no system exists for precisely calculating the design from the operating parameters.

It was necessary to develop a practical model that could be used to extrapolate existing designs to meet different specifications. However in order to do this detailed knowledge of the dynamic operating parameters of this type of cryocooler was required.

The first stage of the program has been to fully instrument a refrigerator so that various dynamic parameters could be measured.

The second stage of the program will involve the application of these measurements to the design and optimisation of a range of coolers.

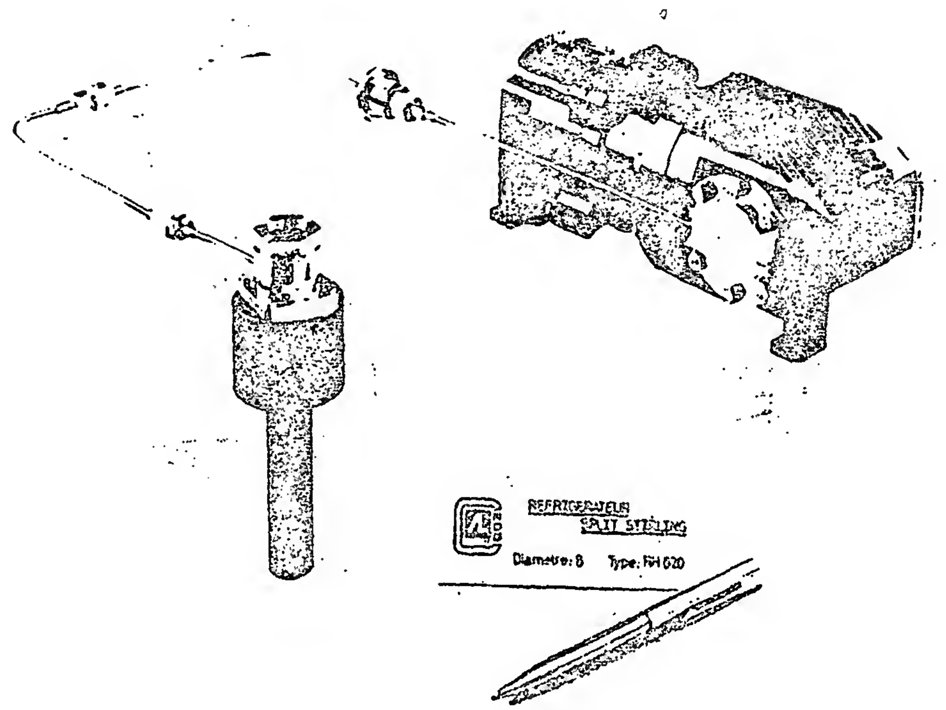
Key words : cryogenic refrigerator, cryocoolers split stirling coolers, refrigeration cycle calculation model, displacer, regenerator, pneumatic displacer drive.

1 - Introduction

L'Air Liquide has been developing and producing miniature cryocoolers for the last ten years. Most of this work has been aimed at products mainly intended for military applications.

In addition to the range of open cycle coolers and flash coolers (ambient to 80 K in about one second), there is an increasing demand for coolers which can operate virtually continuously, and it was to meet this demand that L'Air Liquide undertook the development of split stirling cryocoolers.

As a result of this work an optimised split stirling cryocooler has been designed and built . This machine is shown figure.1.



A major development program was necessary in order to optimise :

- the reliability of the cooler and
- the efficiency of the refrigeration cycle.

In order to improve relisbility, considerable effort has been devoted to the development and selection of technologies to minimise

- leaks
- wear due to friction
- pollution caused by the compressor lubricants.

The estimated without maintenance operating time, has been progressively increased from 500 h to 2000 h (actual test results range from 3000 h to 6000 h).

In parallel, the efficiency of the refrigeration cycle has been optimised for given and fixed values of various system parameters. These parameters include :

- thermal mass to be cooled,
- physical size of the dewar,
- thermal losses in the dewar and
- distance between the dewar and the compressor

Some of the resulting operating characteristics are shown figure 2.

It is evident that for other applications these parameters, and to some extent the specifications (environment temperatures, cooldown time...), may very significantly. Considerable effort would be required to extrapolate and optimise the existing design for other applications. Consequently a new development program was undertaken, concentrating on the development of a design model which would permit a range of cryocoolers to be optimised to meet specific requirements.

Figure 2. L'AL RH 820 split stirling cryocooler caracteristics

Nominal data	
base temperature	76 K (-197°C)
operating temperature range	- 40 °C + 70°C
cooling capacity at 76 K	150 - 1000 mW at 20 °C 150 - 800 mW at 70 °C
Cooldown time	4 min
power supply	20 - 30 V dc
power absorbed	60 W at maximum cooling capacity 25 W with a 200 mW heat load.
system reliability	2000 operational hours (without any maintenance or gas purging)

2 - Initial design criteria

2.1. Method selection

Various starting points were considered, as there already exist different theoritical and practical operating models for the Split Stirling cycle. Such models vary from the extremely simple where, for instance the cycle is assumed to be isothermal, to the vastly complex, where each part of the cycle is modeled in detail.

The first type of model is not sufficiently accurate to optimise a practical machine and the second requires the use of complex calculations and calculating methods, access to which is not always easily available.

Moreover, in order to verify any of the existing models it is necessary to compare the peformance they forecast with the performance and results obtained from real machine, specifically in the case of the split stirling cycle, where the phase shift between pressure and displacement of the piston, and consequently the performance, depend greatly on technological problems (friction, leaks etc...).The corresponding parameters can only be determined by experiment.

In order to do this one must have a system which can be used to measure all the relevant cycle parameters during the operation. Two experimental aproaches can be considered

- either build specific test facilities and measure the different parameters (pressure drop, leaks, friction loads, efficiency) under static conditions.
- or equip a real machine with different instruments and use it to measure the same or equivalent parameters under actual working conditions i.e. under variable cyclic pressure, temperature, volume etc...

The first method is the simplest way to handle the measurement problems.

However, it requires the construction of several specific test assemblies and the extrapolation of experimental results to actual cryocooler performance needs elaborate calculation. It will also include approximations resulting from discrepancies between test conditions and operating conditions. This method seems well adapted to the development of a rather multi-purpose model for the prediction of performance for a wide range of machines.

The second method, i.e. measurement of internal parameters on an operating machine, presents technological difficulties due to the small size of the cryocooler subassemblies and to the high rate of change of the parameters concerned. Provided the associated problems can be solved, this method enables the definition of a rather simpler model which is well adapted to the sizing of similar machines.

In view of our specific requirements, this second method has been selected. Our efforts were concentrated on the selection and development of the measurement methods and the measuring equipment.

2.2. Operating procedure

Generaly speaking the starting point for cryocooler sizing is the power requirement at the end of the cold finger. This nett cooling power results from the difference between.

- the cold power generated during each cycle by the gas expansion or $\int PdV$ where P is the pressure at the cold tip and dV the variation of the cold volume and
- the thermal losses (heat conduction, shuttle losses...).

PV diagram recording and simultaneous measurement of the nett cooling power enable approximation of the total losses and the verification of the corresponding values calculated from the model.

In the same way, instantaneous measurements of pressure, temperature and volume at both, cold and warm ends of the cold finger can be used to calculate regenerator efficiency, pressure drop, etc...and to establish the correlation between the thermodynamic gas cycles at both ends of the finger.

Identical measurements in the compressor cylinder(s), the compressor crank case and the pneumatic spring volume of the cold finger give information which can be used to determine the total pneumatic power absorbed by the gas and the different losses due to

- the gas leak from the compressor cylinder to the crankcase
- the gas leak in the pneumatic spring
- the pressure drop in the connecting line

A power balance in the compressor (friction losses, motor efficiency, power supply and electronic efficiency) gives the required electrical power.

In parallel the force balance in the pneumatic spring allows a comparison between the forecast and actual displacement of the piston to be made thus determining the sizing of pneumatic drive .

With this step by step method, based on the previous experimental optimisation of a particular cryocooler for specific operating conditions, cryocoolers can be designed for differents conditions corresponding to other specifications.

However, in order to do thie the following parameters must be meaeured end recorded :

- the pressure end the tempereture at both cold end werm ends of the cold finger
- the pressure in the compreseor cylinder(s), the pneumetic spring end the compreesor crankcese.
- the poeition of the regeneretor-displacer piston end of the compressor piston(e)
- the ges flow in the connecting line

Thie edds up to a totel of 10 operating perameters.

3 - Experimentel errengement :

3.1 - Test bed :

A RH 800 cryocooler hes been used, which is the first development version of the present standerd RH 820 unit. The RH 800 has the following differences in comparison with the present machine

- single cylinder compressor (The R H 820 is a flat twin compressor to reduce vibration)
- No temperature control,(the RH 820 model is squipped with a cold end temperature controller to monitor rotationel speed end thus cold power with respect to actuel dewar heat losses when opersting over the required - - 40°C to 70°C temperature renge).

The rotationel speed cen be adjusted from 800 to 1500 RPM. Apart from rotationel speed other opereting parameters can be adjusted :

- the mean pressure (from 9 to 25 bar)
- the connecting rod diameter of the pneumatic displacer spring
- the cold end temperature

A heater in the test Dewar allowed the temperature to be set at the desired level.

3.2. Operating method :

The refrigerator wae equipped with the measuring instruments mentioned above which had to be adapted to cope with the small size end to speciel operating requirements, such es

- no increase of dead volume (pressure)
- no increase in heat loss (temperature)
- no perturbation of dynemic balence (displacement)
- high frequency response (cycle cen lest 50 ms and less)
- small dimensions

3.2.1. Pressure measurement :

The pressure sensor must meet the following requirements :

- minute size (sensor diameter 1,5 mm)
- no dead volume (less than a few mm3)
- frequency response 10 kHz
- high offset : small pressure variations at a high mean pressure (in crank case end pneumatic spring)
- low temperature operation (down to 50 K at cold end)

The two last requiremente become increasingly difficult when the first ones are met (problems of the differential pressure, sensor output verietions due to large tempereture differences).

3.2.2. Temperature measurement :

Whilst wall temperature measurements at the warm end present no specific problems this is not the case for the gas temperature measurements, specifically in the cold expansion volume. Thermal inertia requirements leads to the choice of resistance wire of 1 to 5 μm.

The compromise between power input and temperature resolution at cold end has been difficult to reach.

3.2.3. Displacement measurement

The small volume available, both on the cold finger and on the compressor, as well as the problem of not disturbing the equilibrium of the system, have limited the choice to optical and Eddy current sensors. Availibity problems with these measuring sensors have delayed the optical measurements.

The Eddy current sensors have been installed perpendicular to the axis of the piston movements and measure the displacement/distance by means of an inclined groove in the piston, which is machined between the piston rings.

3.2.4. Flow measurement :

Though difficult, a hot wire flow meter was installed in a 1 mm diameter tube, allowing the flow measurements to be made.

3.2.5. Data recording

The rapidly changing parameters required the use of a high performance, real time data logger to achieve high resolution (better than 1 %) at a rate of more than 100 samples per cycle for 10 measurements, some of which are low level.

3.2.6. Power measurement

A calorimetric bench was developped to measure the power balance in the compresor module. The compressor module was immersed in a boiling Freon bath. The freon vapor from the bath was condensed and weighed in order to measure the heat input to the bath.

Comparison with electrical input and PV measurements in the compressor cylinder gives an estimate of
- motor inefficiency (joule losses, eddy current losses)
- friction losses
- non isentropic compression

4. - First experimental results

A considerable amount of development work was required due to problems encountered with these type of measurement.

Solutions have been found and several hundred sets of parameters have been tested, each set consisting of :
- mean pressure
- compressor motor revolution speed
- connecting rod diameter for displacer
- regenerator diameter in the displacer
- cold end temperature

Some exemples of the recorded data are shown in figure 3.

The diagrams 3a and 3b represent an optimal situation. The PV diagram is virtually rectangular. The displacer moves exactly at the maximum and minimum pressures.

The diagrams 3c and 3d represent a non-optimised situation

A shape coefficient has been defined in order to measure the level of optimization .

The variations of the shape coefficient as a function of the parameters listed above and the influence of the connecting rod diameter of the displacer in connection with the other parameters are shown in figure 4.

This figure emphasizes the importance of pneumatic drive sizing in order to have the proper phase shift and thus the maximum cold power.

In addition it illustrates the difficulty in optimising a cryocooler for a wide operating range. A cryocoler can be optimised for the highest frequency and the maximum pressure and hence have the proper phasing at maximum cold power output resulting in the shortest cool down time and, non-optimised performance under nominal conditions or vice versa.

For instance a cool down time of less than 2 minutes has been obtained with a particular design, at the cost of lower specific performance under steady state conditions.

5 - Conclusion

The first steps in experimental modelisation of the split stirling cycle have been made. Having solved the delicate measurement problems a large amount of experimental data has become available and now awaits processing. The experiments described above give a better insight into the cycle and a better understanding of the various parameters involved with the dimensioning of cryocoolers.

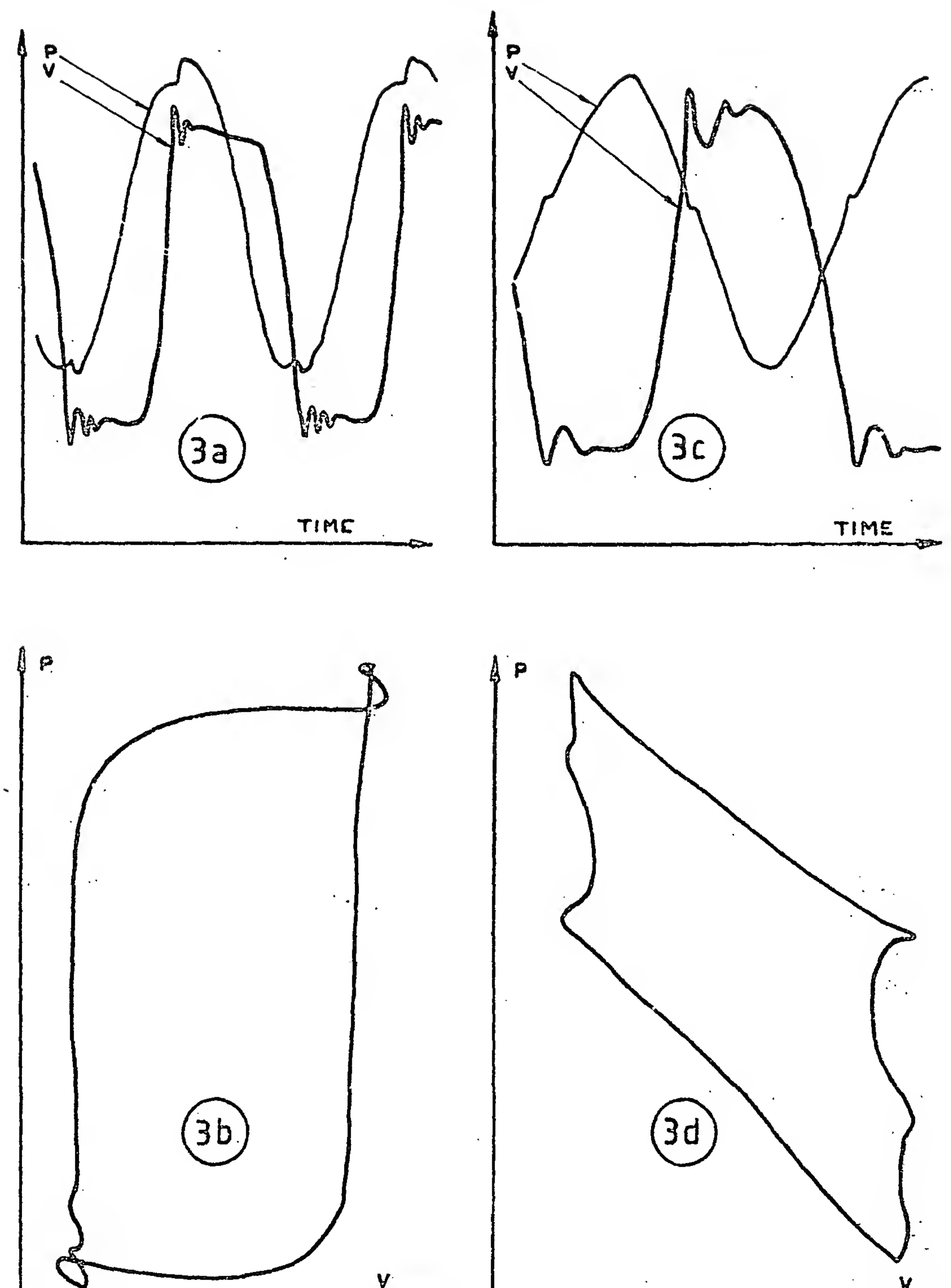
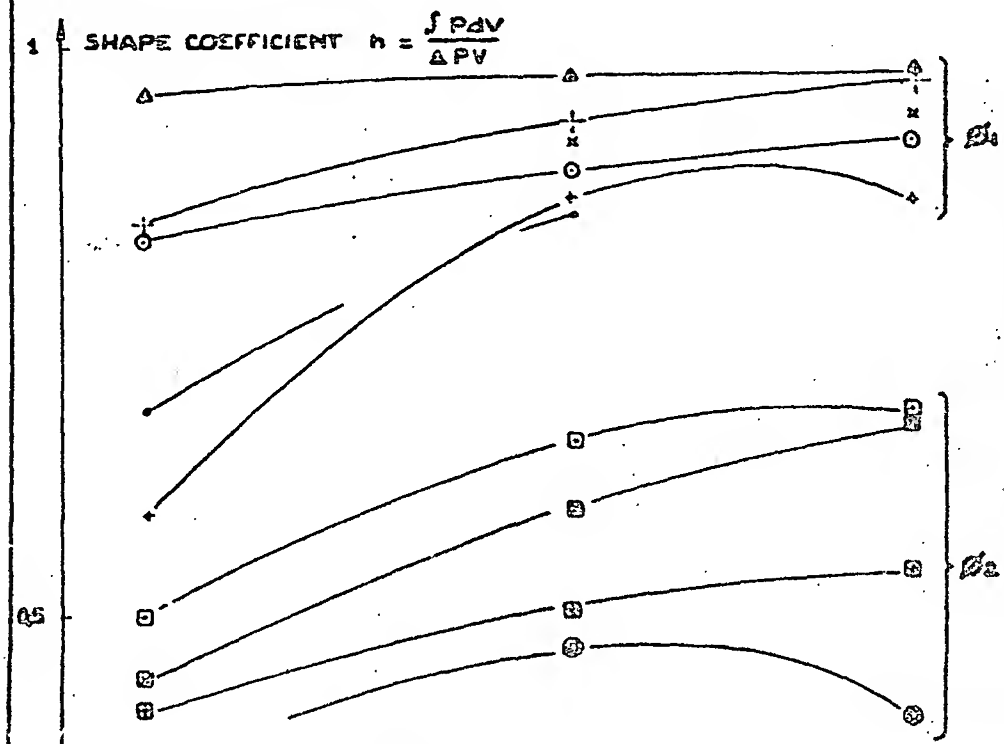


FIG.3 PV DIAGRAM AT THE COLD END

3a.3b OPTIMISED CONFIGURATION
3c.3d NON OPTIMISED CONFIGURATION



	Ø1			Ø2		
P / f	9	15	20	9	15	20
13,3			△			•
20	+	×	+	⊛	⊟	⊡
25			⊙		⊞	

-197 -120 -60

COLD END TEMPERATURE °C

P : AVERAGE PRESSURE (BAR)
f : FREQUENCY (Hz)
Ø : PNEUMATIC DRIVE ROD DIAMETER

FIG.4 VARIATIONS OF THE PV DIAGRAM AREA